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COLD-AIR INVESTIGATION OF A TURBINE WITH TRANSPIRATION-COOLED STATOR BLADES

II - Stage Performance With Discrete Hole Stator Blades

by Edward M. Szanca, Harold J. Schum, and Frank P. Behning

Lewis Research Center
Cleveland, Ohio 44135

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16. Abstract

Turbine performance characteristics were obtained for a single-stage axial-flow turbine equipped with stator blades employing transpiration coolant ejection through discrete holes. The turbine was tested over a range of speed and pressure ratio, and with a coolant fraction of 0.03. Additional tests were made at the design speed in which the coolant flow was varied from zero to 0.07 of the primary flow. The results were compared to similar results obtained with a turbine using stator trailing-edge coolant ejection. A base turbine with no cooling provision was used as a standard for comparisons.

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SUMMARY

A cold-air experimental investigation was conducted on a 30-inch (0.762-m) single-stage turbine equipped with transpiration-cooled stator blades having coolant ejection through discrete holes. The investigation was made to determine the effect of coolant flow ejection through this type of blade on turbine aerodynamic performance. The effect of coolant on turbine performance was determined by testing over a range of coolant fraction (ratio of coolant flow to primary flow) from zero to 0.07.

The performance of this turbine was compared to a turbine with stator trailing-edge coolant ejection and also to a base reference turbine with no coolant flow provision. The stator blades for all three turbines were of the same profile geometry, and were tested with the same rotor.

At a coolant inlet supply pressure of 31 inches of mercury absolute (10.498 N/cm^2), equivalent design speed, and at an equivalent specific turbine work output of 17.00 Btu per pound (39 572 J/kg), the primary air efficiency for the transpiration-cooled turbine was 0.907. The corresponding overall pressure ratio was 1.771 and the coolant fraction 0.0326. At this speed and work output, the turbine primary efficiency increased slightly with increasing coolant fraction. The thermodynamic efficiency decreased in almost direct proportion to the coolant fraction as a result of a degradation in both stator and rotor performance.

The primary and thermodynamic efficiencies of the transpiration-cooled turbine were both significantly less than the corresponding efficiencies obtained with the turbine employing stator-trailing-edge coolant ejection. For example, at the same work output and speed as mentioned previously, and at a coolant fraction of 0.07, the thermodynamic efficiency of the transpiration-cooled turbine was 0.09 less than at a coolant fraction of zero. In contrast, the thermodynamic efficiency with trailing-edge coolant ejection remained essentially constant.

INTRODUCTION

The performance requirements of gas turbine engines currently being considered for advanced types of aircraft engines necessitate high turbine inlet temperatures. These high temperatures require turbine blade cooling. The effect of this cooling air on the aerodynamic performance of the turbine must be clearly understood in order to be factored into the turbine design and engine performance assessment. Accordingly, various stator blade cooling schemes are being investigated at NASA Lewis Research Center. Two of these are stator trailing-edge coolant ejection and transpiration cooling with discrete holes.

To incorporate these cooling schemes, a 30-inch (0.762-m) cold-air research turbine was designed with physical features suitable for blade cooling. The turbine blading for this type of turbine is characterized by thick profiles and blunt leading and trailing edges. The design procedure, together with coordinates for the stator and rotor blade profiles, is presented in reference 1. The performance of the reference turbine is reported in reference 2. In addition, the stator was tested as a separate component, and a detailed stator loss determination is presented in reference 3. This turbine will hereinafter be referred to as the "base" turbine.

The stator blades of the base turbine were then modified to incorporate a coolant air ejection slot along the trailing edge of the stator blade. The coolant air was ejected into the main stream along the entire radial length of the blade and in the same general direction as the primary air. Tests were conducted to determine the effect of this type of coolant ejection on turbine performance. The results of these tests, which are reported in references 4 to 6, show that the modification of the trailing edge and coolant ejection had very little effect on either stator or turbine thermodynamic efficiency. From these results, it was also concluded that the rotor efficiency was not significantly affected by the coolant flow.

A second cooled configuration was then built using a transpiration-type cooling scheme in the stator, maintaining the same profile as for the base turbine. In this cooling scheme, the coolant flow was ejected through holes in the entire blade surface and in a direction normal to the primary flow. The blade is a self-supporting shell with a single chamber which supplies the coolant flow through discrete holes around the blade surface. The results of the stator component tests employing this type of blade are reported in reference 7. In that report it was shown that stator performance significantly decreased when this transpiration cooling method was used.

The purpose of the investigation reported herein was to experimentally determine the effect of these transpiration-cooled stator blades on turbine aerodynamic performance. Performance parameters in terms of mass flow, torque, and efficiency were obtained over a range of speed and pressure ratio for a nominal coolant fraction (ratio of coolant flow to primary flow) of 0.03. Also, the effect of varying the coolant fraction was determined over a limited range of pressure ratio at the design speed. The coolant fraction investigated ranged from zero to 0.07. The performance of this transpiration-cooled turbine is compared with the performance of both the stator-trailing-edge-cooled turbine and the base turbine.

All turbine tests were conducted at a constant inlet stagnation pressure of 30.0 inches of mercury absolute (10.159 N/cm^2). Both the primary and coolant air was supplied by the laboratory combustion air system at a nominal temperature of 545° R (303 K). Hence, no heat-transfer effects were considered.

SYMBOLS

A	annular flow area, ft^2; m^2
g	force-mass conversion constant, 32.174 ft/sec^2
h	specific enthalpy, Btu/lb; J/kg
J	mechanical equivalent of heat, 778.16 ft-lb/Btu
N	rotational speed, rpm
p	absolute pressure, lb/ft^2; N/m^2
R	gas constant, 53.34 ft-lb/(lb)(°R); 287 J/(kg)(K)
T	temperature, °R; K
U_m	blade velocity at mean height, ft/sec; m/sec
V	absolute gas velocity, ft/sec; m/sec
w	mass-flow rate, lb/sec; kg/sec
α	absolute flow angle, measured from axial, positive in direction of rotor rotation, deg
γ	ratio of specific heats
δ	ratio of turbine inlet total pressure to U.S. standard sea-level pressure of 29.92 in. Hg abs (10.132 N/cm^2)
η	efficiency based on total-pressure ratio
$\sqrt{\theta_{cr}}$	ratio of critical velocity at turbine inlet to critical velocity at U.S. standard sea-level air temperature of 518.7° R (288 K)
τ	torque, ft-lb; N-m

Subscripts:

a actual

c coolant flow

cr conditions at Mach 1

h hub radius

id ideal

p primary flow

r rotor

t tip radius

th thermodynamic

0 measuring station at turbine inlet (see fig. 5)

1 measuring station at stator outlet

2 measuring station at rotor outlet

Superscript:

' total state

APPARATUS AND INSTRUMENTATION

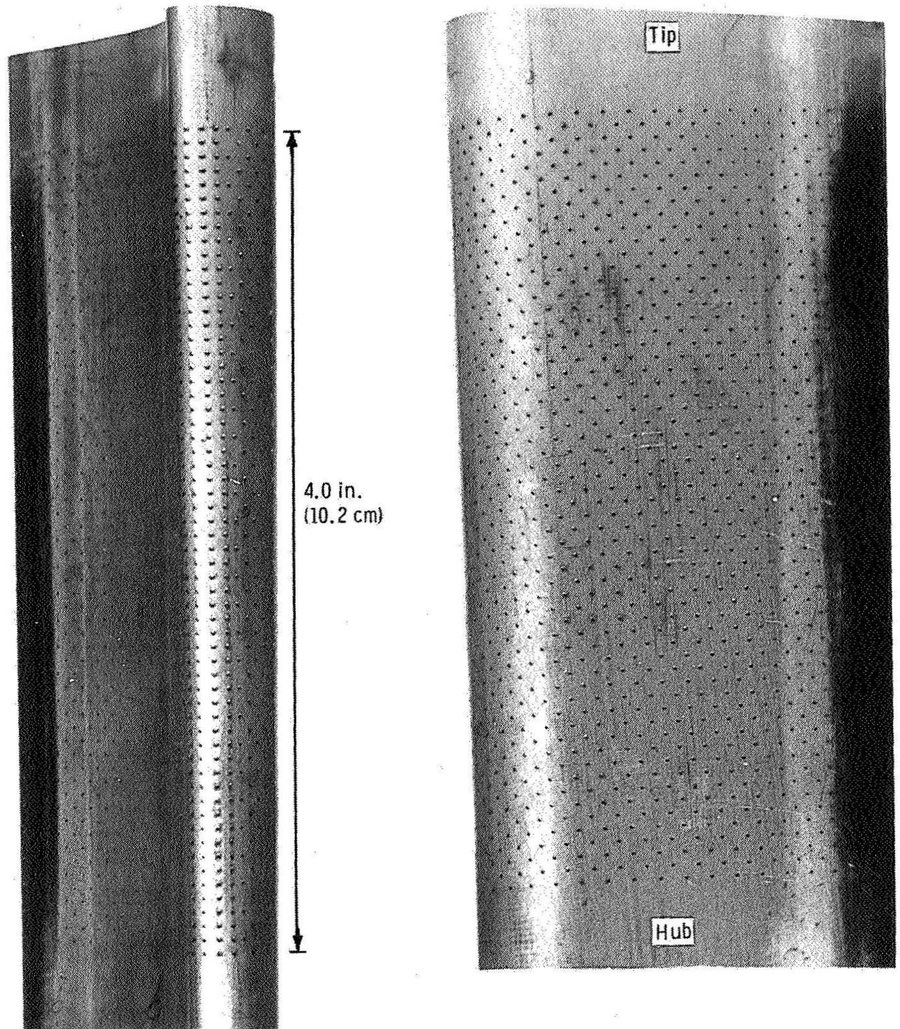
The turbine used in this investigation was a 30-inch (0.762-m) tip diameter, single-stage, axial-flow turbine of the same aerodynamic design as the base turbine reported in reference 1. The design requirements of the base turbine are summarized as follows:

Equivalent specific work output, $\Delta h/\theta_{cr}$, Btu/lb (J/kg) 17.00 (39 572)

Equivalent mean blade speed, $U_m/\sqrt{\theta_{cr}}$, ft/sec (m/sec) 500 (152.4)

Equivalent mass flow, $w\sqrt{\theta_{cr}}/\delta$, lb/sec (kg/sec) 39.9 (18.098)

The design procedure used to evolve the blade shapes and a sketch showing the blade passage and profiles and the blading coordinates can also be found in reference 1. The stator blades of the reference turbine, however, were replaced with stator blades constructed of a porous material with discrete holes normal to the blade surface. These blades are of a hollow construction with no internal supports or flow passages. Coolant flow is controlled by a combination of the blade cavity pressure, the size and spacing of the circular apertures on the surface of the blade, and the pressure distribution along the surface of the blade. Closeup views of the blade are shown in figure 1, and the



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(a) Leading edge and pressure surface.

(b) Suction surface.

Figure 1. - Transpiration-cooled stator blades.

Figure 2. - Stator assembly.

blades installed in the stator ring are shown in figure 2.

The turbine rotor used in this performance evaluation was the same rotor used for the base turbine described in reference 2, and for the stator-trailing-edge-slot investigation reported in reference 6. The rotor is shown in figure 3.

The test facility, described in reference 2, was modified to incorporate a stator cooling air system as described in reference 6. This system included an air supply pipe from the combustion air header, an airflow venturi meter, a control valve, the coolant inlet torus, the coolant supply annulus, and appropriate instrumentation for determining the coolant mass-flow rate. The inlet torus was connected to the supply annulus by twelve $1\frac{1}{2}$-inch (3.81-cm) feeder pipes, which, in conjunction with a circumferential baffle, distributed the air evenly over the outside circumference of the stator. Both the coolant and primary air were supplied by the laboratory combustion air system. The test facility is shown in figure 4.

The instrumentation was essentially the same as described in reference 2 except for that required to determine the coolant mass flow rate and its ideal energy. This instrumentation measured the venturi upstream pressure, the differential pressure, and

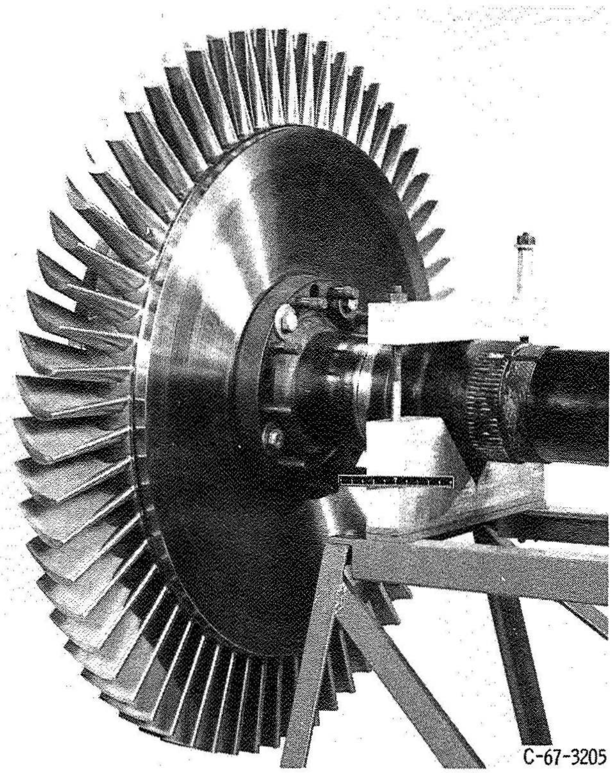

Figure 3. - Turbine rotor assembly.

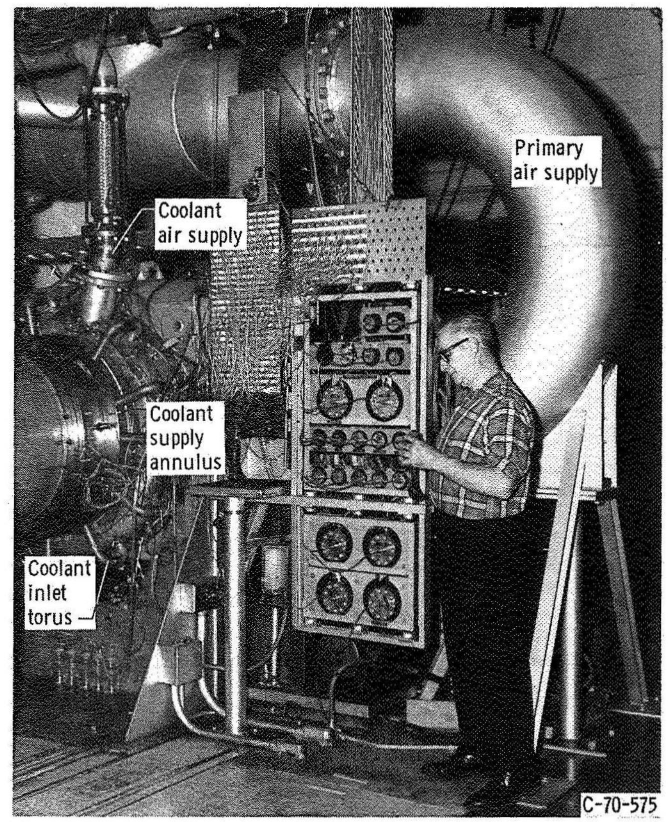

Figure 4. - Test facility.

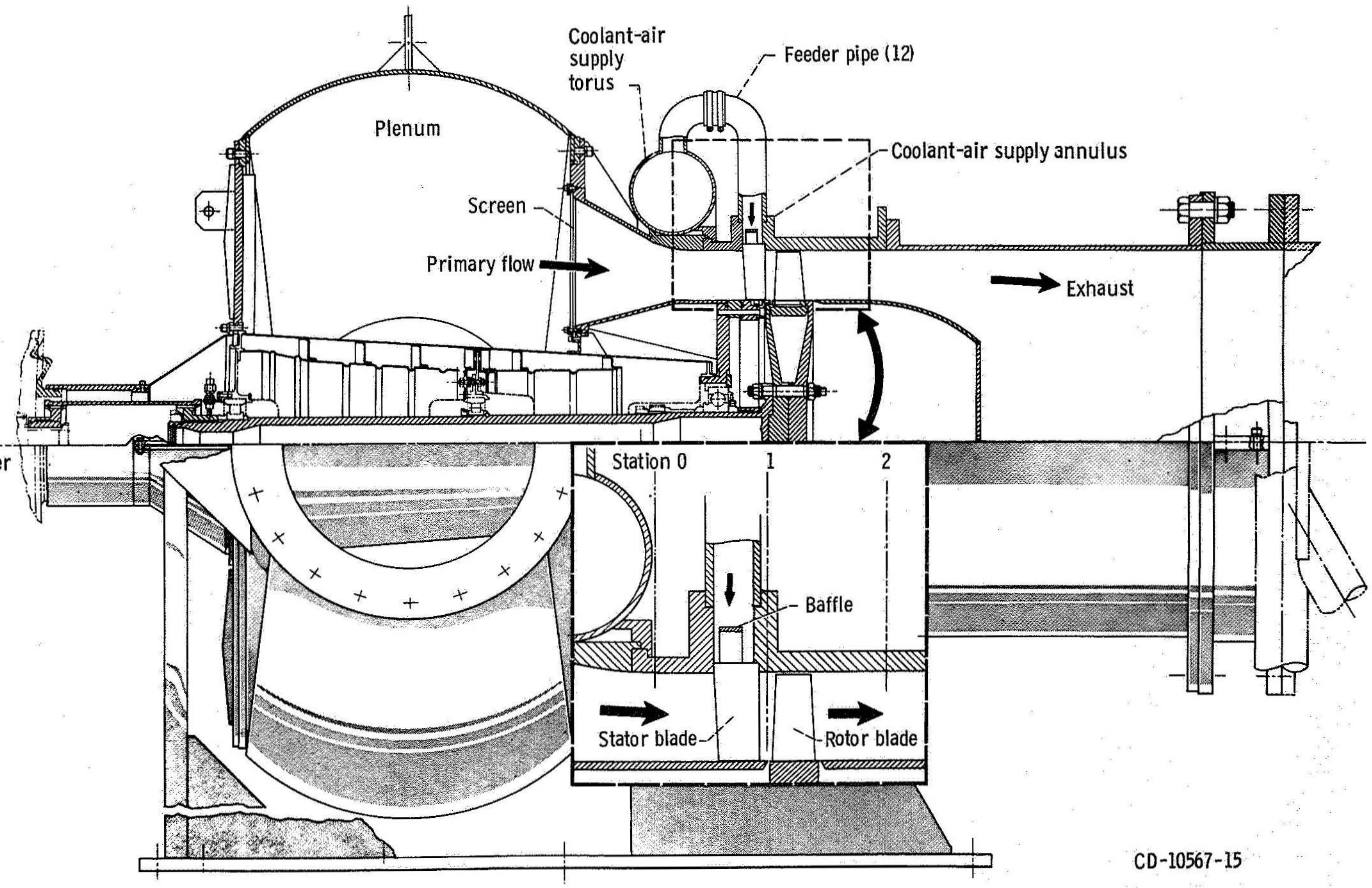

Figure 5. - Turbine test section.

the coolant-air temperature. Instrumentation was also provided to measure the temperature and pressure of the coolant at the coolant supply annulus.

The other instrumentation used was described in detail in reference 2 and measured the following: total and static pressures and temperatures at the turbine inlet, static pressures at the stator exit, turbine exit static pressure, turbine exit flow angles, turbine speed, and torque. The measuring stations are shown in figure 5.

All instrumentation was connected to a 100-channel data acquisition system which measured and recorded the electrical signals from the appropriate transducers. At each data point, five readings of each transducer were recorded and subsequently numerically averaged.

PROCEDURE

The performance tests for this transpiration-cooled turbine were conducted in two phases. In phase one, the performance data were taken over a range of overall pressure ratio and speed with the cooling air supply annulus pressure equal to 31 inches of mercury absolute (10.498 N/cm^2). The coolant supply annulus pressure was maintained slightly higher than the turbine inlet pressure (30.0 in. Hg abs; 10.159 N/cm^2) in order to ensure a positive pressure drop from the blade cavity to the blade surroundings. The speed was varied over a range of from 60 to 110 percent of equivalent design speed (4407 rpm) in 10-percent increments. At a given speed, the pressure ratio was varied from approximately 1.4 to pressure ratios into the choked-flow region.

In phase two, performance data were taken at design speed over a range of coolant fraction and for pressure ratios bracketing a work output of 17.00 Btu per pound (39 572 J/kg). The coolant mass-flow rate was varied by regulating the pressure in the coolant supply annulus. The variation of this pressure resulted in blade cavity pressures both below and above the turbine inlet pressure for the range of coolant fractions reported. The range of coolant fraction investigated was from zero to 0.07.

In both phases, the turbine inlet primary air, total-state conditions were 30.0 inches of mercury absolute (10.159 N/cm^2) and approximately 545° R (303 K). Overall pressure ratios were set by adjusting the turbine exit pressure.

Turbine performance was based on total-pressure ratio. The inlet total pressure was calculated (as in refs. 2 and 6) from the static pressure, primary mass flow, annulus area, and total temperature by using the following equation:

$$\frac{p_0'}{p_0} = \left[\frac{1}{2} + \sqrt{\frac{1}{4} + \frac{\gamma - 1}{2g\gamma}\left(\frac{w_p}{p_0 A_0}\right)^2 RT_0'}\right]^{\gamma/(\gamma-1)} \tag{1}$$

The outlet total pressure ratio was calculated (as in refs. 2 and 6) by using static pressure, combined primary and coolant mass flows, annulus area, area-averaged turbine exit flow angle, and total temperature, as follows:

$$\frac{p_2'}{p_2} = \left[\frac{1}{2} + \sqrt{\frac{1}{4} + \frac{\gamma - 1}{2g\gamma}\left(\frac{w_p + w_c}{p_2 A_2}\right)^2 \frac{RT_2'}{\cos^2 \alpha_2}}\right]^{\gamma/(\gamma-1)} \tag{2}$$

The total temperature used in equation (2) was derived by using the inlet total temperature, torque output, total mass flow, and speed data.

Two efficiencies are defined for use in this report: the primary efficiency η_p and the thermodynamic efficiency η_{th}. In equation form,

$$\eta_p = \frac{w_p \Delta h_p + w_c \Delta h_c}{w_p \Delta h_{id,p}} = \frac{\frac{\tau N \pi}{30J}}{w_p \Delta h_{id,p}} \tag{3}$$

$$\eta_{th} = \frac{w_p \Delta h_p + w_c \Delta h_c}{w_p \Delta h_{id,p} + w_c \Delta h_{id,c}} = \frac{\frac{\tau N \pi}{30J}}{w_p \Delta h_{id,p} + w_c \Delta h_{id,c}} \tag{4}$$

The primary efficiency, which relates the total power of both fluids to the ideal power of only the primary flow, is useful in engine cycle studies. The thermodynamic efficiency, which accounts for the ideal energies of both fluids, is useful in studying the loss characteristics of the fluids involved.

RESULTS AND DISCUSSION

The results of this experimental investigation are discussed in two parts. First, the overall turbine performance with cooling air supplied at a coolant annulus pressure 1 inch of mercury (0.339 N/cm^2) above the turbine inlet pressure is discussed. These results are then compared to those previously obtained for both the turbine with the slotted-trailing-edge stator configuration (ref. 6) and the base turbine (ref. 2).

Second, the effect of coolant flow on the performance of the transpiration-cooled turbine at design speed is discussed. The performance of this turbine is compared to the performance of the stator-trailing-edge-ejection-type turbine (ref. 6). The coolant fraction range investigated with both turbines is from zero to 0.07.

Turbine Performance at Constant Coolant Supply Annulus Pressure

Overall turbine performance. - The overall performance map for the transpiration-cooled turbine is presented in figure 6. Turbine performance is presented in terms of equivalent specific work output $\Delta h/\theta_{cr}$ and a mass-flow - speed parameter $w_p N/\delta$, both based on primary flow. Lines of constant total-pressure ratio p_0'/p_2' and equivalent rotor speed $N/\sqrt{\theta_{cr}}$ are indicated. Contours of constant primary efficiency η_p, based on the total-pressure ratio across the turbine, are also included. The coolant fraction over the range of speed and pressure ratio covered by the map was about 0.03.

At the design speed, and at a specific work output of 17.00 Btu per pound (39 572 J/kg), corresponding to the equivalent design work of the base turbine, the efficiency was 0.907. The pressure ratio corresponding to this efficiency was 1.771. In general, high turbine efficiencies were obtained over a broad range of speed and pressure ratio. Efficiencies over 0.90 were obtained at 100 percent speed from the lowest pressure ratio investigated (1.4) to a pressure ratio of slightly over 2.1. Maximum efficiencies of over 0.915 occurred at 110 percent of design speed over a range of pressure ratio from 1.60 to 1.75.

Torque and mass-flow characteristics. - The cooled-turbine performance map (fig. 6) was evolved from the torque data τ/δ of figure 7 and the primary mass-flow data $w_p\sqrt{\theta_{cr}}/\delta$ of figure 8. Both the torque and mass flow are presented in equivalent terms and plotted as functions of overall total-pressure ratio and speed. The torque curves show that the torque increased with increasing pressure ratio for all speeds investigated. For any constant pressure ratio, the torque increased with decreasing speed. Limiting blade loading, defined as that point where increases in overall pressure ratio result in no increase in torque output, was not attained at any speed.

The primary mass-flow curves (fig. 8) show increasing flow with increasing pressure ratio for all speeds, until choked (constant) values are reached. The separation of the various speed lines in the choked range indicates that the flow is being controlled by the rotor. At 60 percent of equivalent design speed, the rotor choked at a total-pressure ratio of about 1.81; and at 110 percent of design speed, the choking pressure ratio was approximately 2.23. At design equivalent speed, a choking mass flow of 41.16 pounds per second (18.67 kg/sec) was obtained for total-pressure ratios above 2.19. At a pressure ratio of 1.771, corresponding to a primary work output of 17.00 Btu per pound (39 572 J/kg) at design speed, the primary mass flow was 40.14 pounds per second (18.21 kg/sec).

Turbine outlet flow angle. - The average turbine outlet flow angle α_2 is shown in figure 9 as a function of total-pressure ratio and speed. The angle shown is the numerical average of the angle measurements taken at radial positions corresponding to the area center radii of five equal annular areas. The negative sign of the angle corresponds

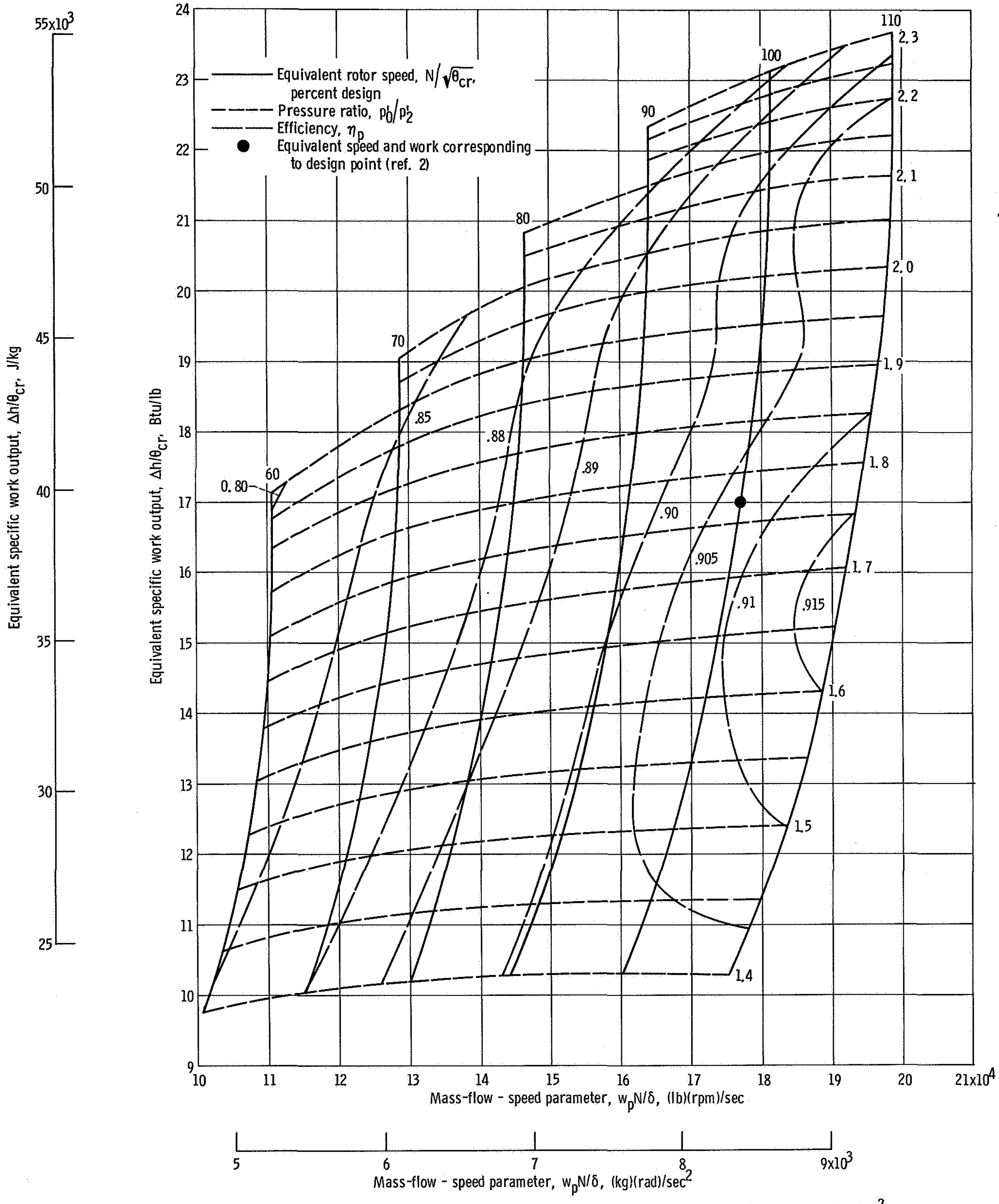

Figure 6. - Overall performance map. Turbine inlet pressure, 30.0 inches of mercury absolute (10.159 N/cm^2); coolant inlet pressure, 31 inches of mercury absolute (10.498 N/cm^2).

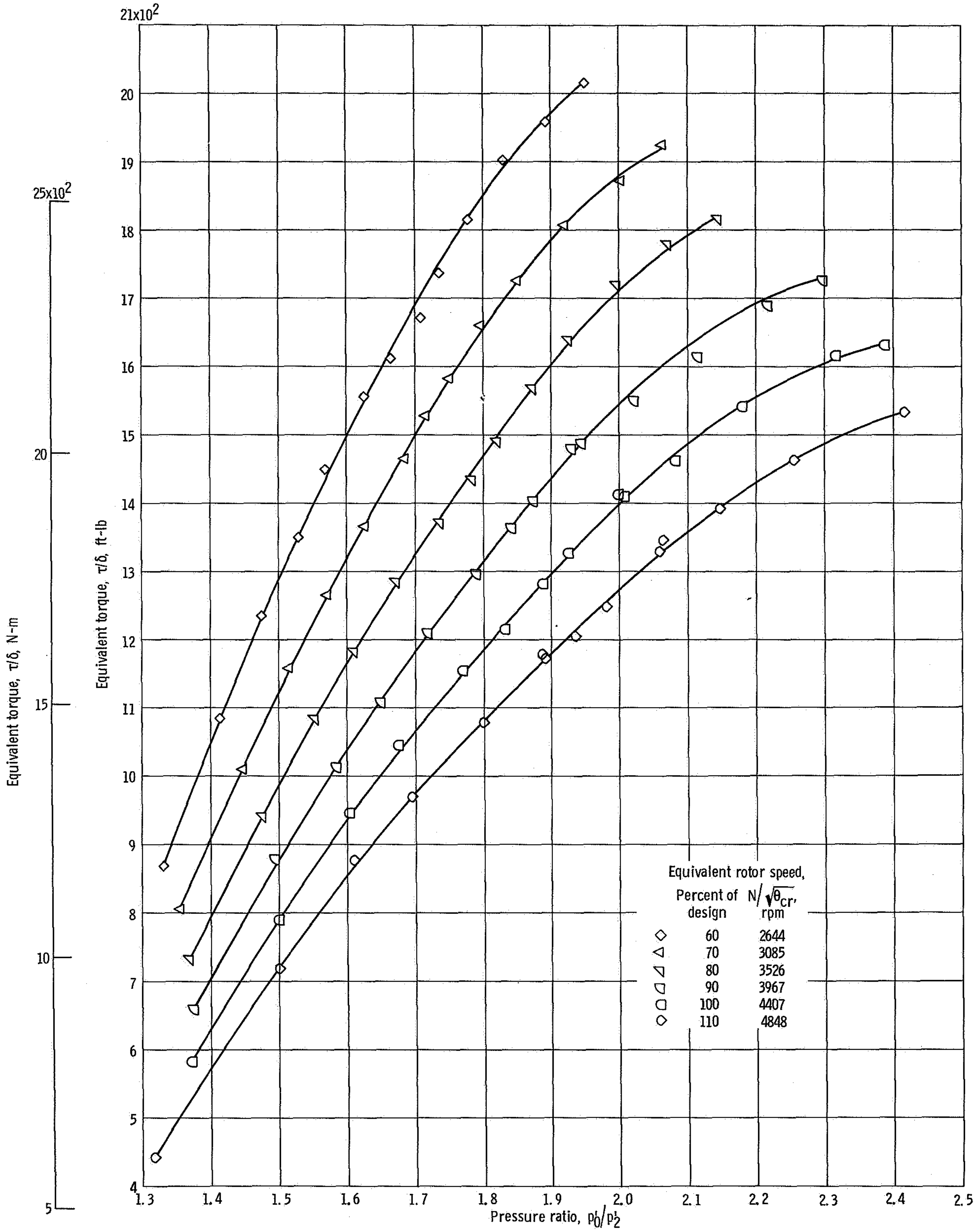

Figure 7. - Variation of equivalent torque with pressure ratio and equivalent speed. Turbine inlet pressure, 30.0 inches of mercury absolute (10.159 N/cm^2); cooling air pressure, 31 inches of mercury absolute (10.498 N/cm^2).

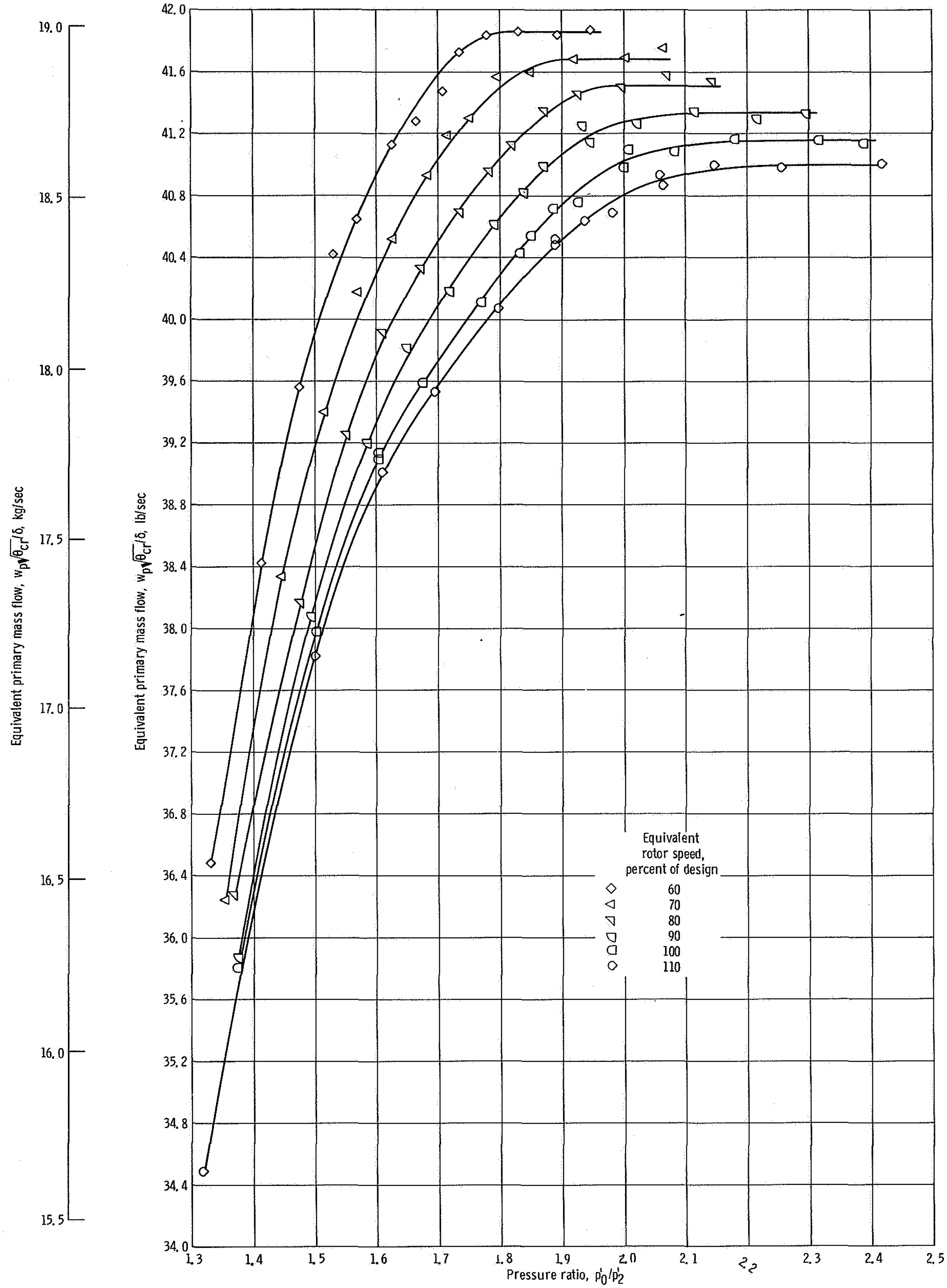

Figure 8. - Variation of equivalent primary mass flow with pressure ratio and equivalent speed. Turbine inlet pressure, 30.0 inches of mercury absolute (10.159 N/cm²); coolant inlet pressure, 31 inches of mercury absolute (10.498 N/cm²).

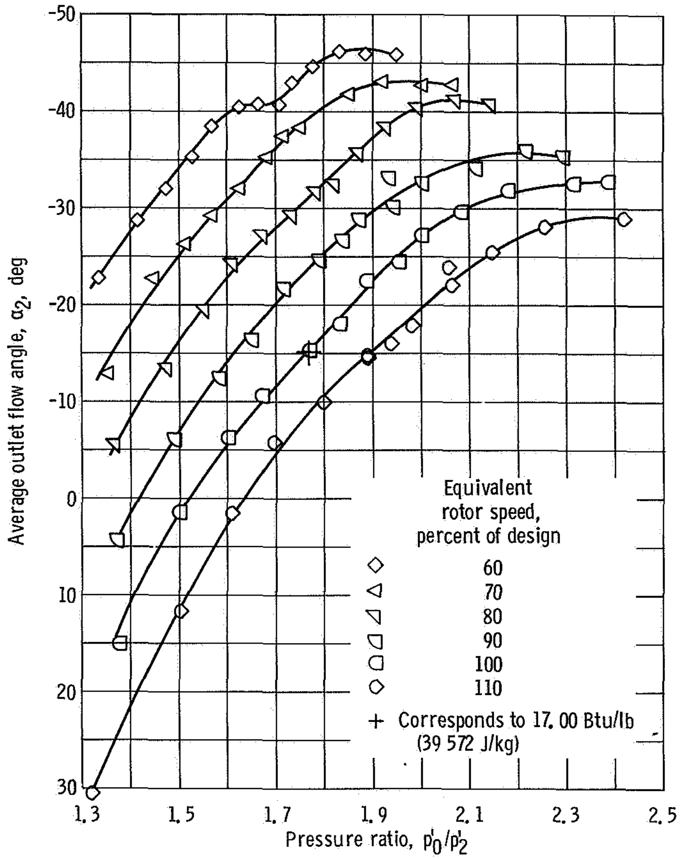

Figure 9. - Variation of turbine outlet flow angle with pressure ratio and equivalent rotor speed. Turbine inlet pressure, 30.0 inches of mercury absolute (10.159 N/cm^2); coolant inlet pressure, 31 inches of mercury absolute (10.498 N/cm^2).

to a positive contribution to the turbine work output. The average angle indicated at conditions of design speed and at a pressure ratio of 1.771 was -15.2°. This corresponds to a specific work output of 17.00 Btu per pound (39 572 J/kg) based on primary flow.

Radial pressure gradient. - The radial pressure gradient from hub to tip at the stator exit is shown in figure 10. This pressure gradient was plotted from turbine tests and compared to the results obtained from the stator component testing of reference 7. The data with the rotor in place were obtained at design speed and over a range of overall total-pressure ratio.

Figure 10 shows the pressure gradient with only the stator is greater than the pressure gradient with the rotor in place. However, the pressure gradient obtained from stator component testing is within 0.03 of the pressure gradient obtained from turbine tests. This difference can be attributed to the interaction between the stator and the rotor.

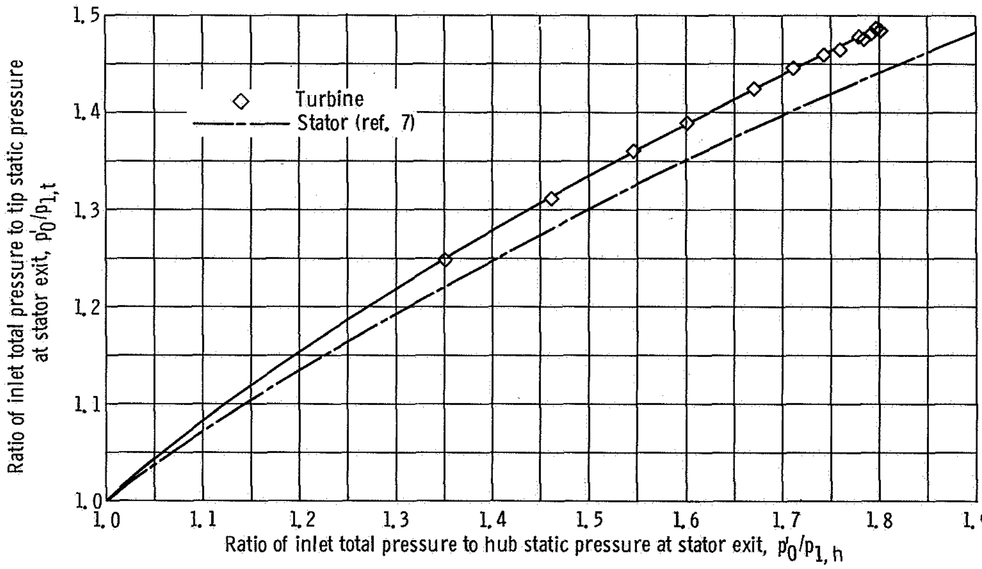

Figure 10. - Radial static-pressure gradient at stator exit for a range of turbine pressure ratio, with and without rotor in place. Turbine speed, 4407 rpm (equivalent design).

TABLE I. - PERFORMANCE COMPARISON OF COOLED TURBINES AND BASE REFERENCE TURBINE AT DESIGN SPEED AND A WORK OUTPUT OF 17.00 BTU PER POUND (39 572 J/kg)

Stator configuration	Equivalent primary mass flow, $w_p\sqrt{\theta_{cr}}/\delta$		Coolant fraction, w_c/w_p	Total-pressure ratio, p_0'/p_2'	Outlet flow angle, α_2, deg	Ratio of coolant supply annulus pressure to turbine inlet pressure, p_c'/p_0'	Primary air efficiency, η_p	Thermo-dynamic efficiency, η_{th}
	lb/sec	kg/sec						
Transpiration cooled	40.14	18.21	0.0326	1.771	-15.2	1.03	0.907	0.876
Trailing-edge ejection (ref. 6)	39.72	18.02	0.0468	1.713	-15.5	1.00	0.958	0.917
Base (design; uncooled, ref. 2)	40.64	18.43	------	1.751	-15.2	----	0.923	0.923

Comparison of cooled and uncooled turbines. - Table I summarizes and compares some of the important turbine performance parameters of the transpiration-cooled turbine to those of the turbine with stator-trailing-edge coolant ejection (ref. 6) and the base design turbine (ref. 2). The performance parameters are tabulated at the equivalent design operating conditions for the base turbine, which were a work output of 17.00 Btu per pound (39 572 J/kg) based on primary flow and a speed of 4407 rpm. Inlet total-state conditions for all three turbines were 30.0 inches of mercury absolute (10.159 N/cm^2) and approximately 545° R (303 K). The coolant supply inlet total-state conditions were approximately equal to the turbine inlet conditions.

The results of table I show no significant change in thermodynamic efficiency between trailing-edge ejection and the base reference turbine. The efficiency of the transpiration-cooled turbine, however, dropped off significantly for both primary and thermodynamic efficiency when compared to the other two turbine configurations.

The differences in efficiency between the two cooled turbines is due primarily to the difference in stator performance resulting from the location and direction of the coolant ejection relative to the primary flow. In reference 7 it was shown that, for design equivalent primary flow, the wakes behind the subject transpiration-cooled stator became thicker and larger as coolant fraction increased. Wakes measured behind the trailing-edge slotted stator for corresponding test conditions were smaller (ref. 5). In fact, above coolant fractions of 0.03, the wakes actually decreased as the coolant fraction was increased.

Effect of Coolant Flow Variation on Turbine Performance

The effect of coolant flow on the transpiration-cooled turbine performance was determined by testing the turbine over a range of coolant fraction and overall total-pressure ratio. The turbine was operated at the design speed and at pressure ratios bracketing an equivalent specific-work output of 17.00 Btu per pound (39 572 J/kg). At each pressure ratio the coolant fraction was varied from zero to approximately 0.07.

Equivalent values of torque and mass flow as functions of total-pressure ratio were evolved from experimental data for incremental values of coolant fraction. These parameters were then used to calculate work output, primary efficiency, and thermodynamic efficiency.

Variations of flow and torque with coolant fraction. - The variation of equivalent primary flow, total flow, and torque, as functions of coolant fraction, is shown in figure 11. The curves are shown for design equivalent speed and a constant overall pressure ratio of 1.778. This pressure ratio corresponds to a work output of 17.00 Btu per pound (39 572 J/kg) at zero coolant fraction. Figure 11 shows that the primary flow through the turbine varies in inverse proportion with the coolant fraction. For example, at a coolant fraction of 0.07, the primary flow was 0.93 of the primary flow at zero coolant fraction.

Figure 11 also shows that the sum of the primary flow and coolant flow (total flow) is relatively constant over the range of coolant fraction investigated. The total flow decreased about 1/2 percent with increasing coolant fraction to about 0.03, and remained essentially constant with further increases in coolant fraction to 0.07.

The equivalent torque (fig. 11) was found to decrease almost linearly with increasing

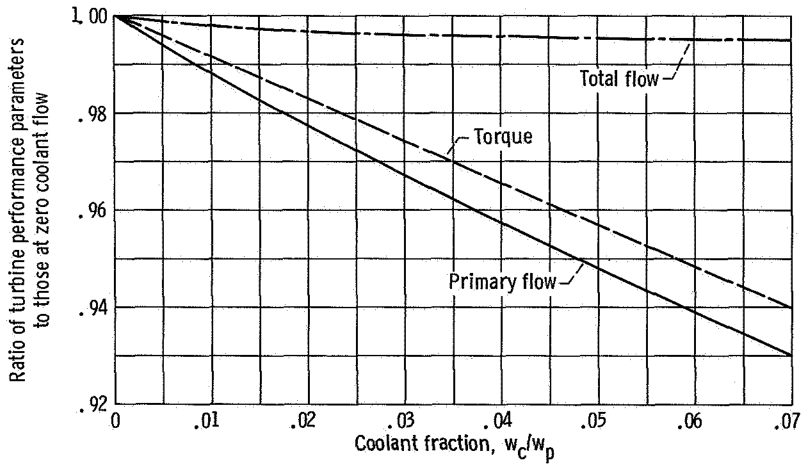

Figure 11. - Variation of equivalent torque, primary flow, and total flow with coolant fraction, at constant overall pressure ratio (1.778) and equivalent design speed.

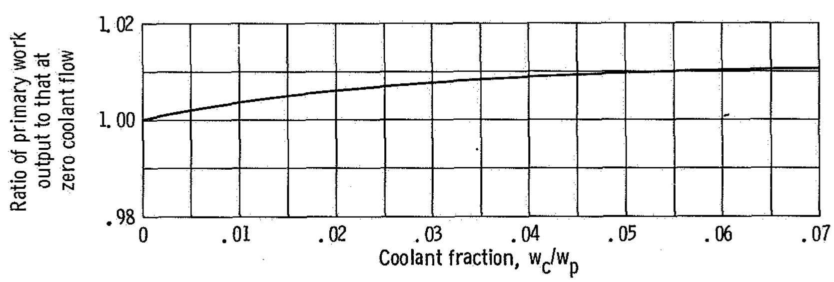

Figure 12. - Variation of primary work output with coolant fraction for equivalent design speed and constant overall pressure ratio. Based on an equivalent work output of 17.00 Btu per pound (39 572 J/kg) at zero coolant fraction.

coolant fraction, although not to the same extent as the primary flow. For example, at a coolant fraction of 0.07, the torque decreased 6 percent as compared to the 7 percent decrease in primary flow.

The primary work output as a function of coolant fraction is shown in figure 12. The work was obtained from the torque and primary flow curves of figure 11, using the relation $\tau N/w_p$. The maximum gain in work output over the range of coolant fraction investigated was about 1 percent.

Variation of turbine efficiencies with coolant fraction. - The variation in thermodynamic and primary efficiencies as functions of coolant fraction for the transpiration-cooled turbine is shown in figure 13. As pointed out in the section PROCEDURE, thermodynamic efficiency is a measure of the loss characteristics when considering both the primary and coolant airflows. The primary efficiency is useful in engine-cycle studies and considers the turbine as being charged only with the ideal energy of the pri-

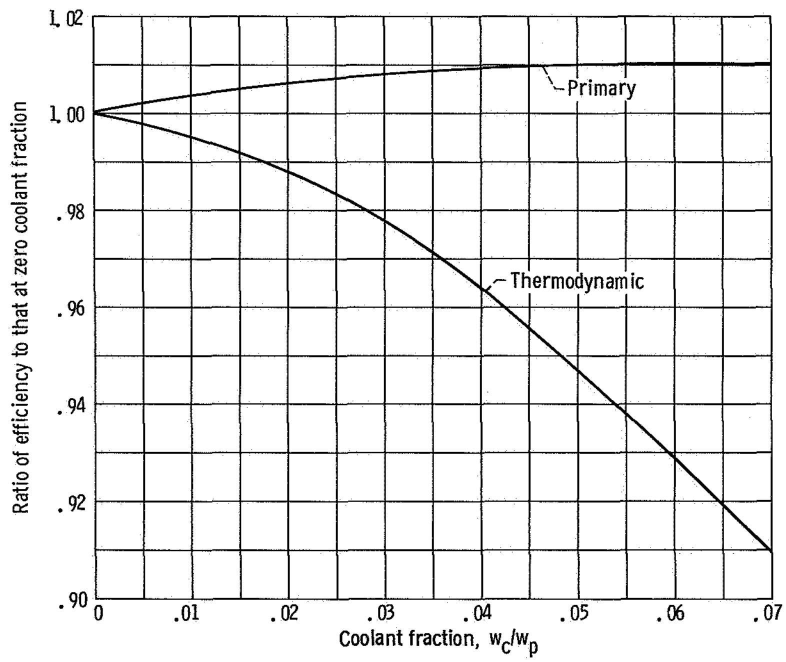

Figure 13. - Variation of efficiency with coolant fraction. Equivalent design speed; equivalent work output, 17.00 Btu per pound (39 572 J/kg), based on primary flow.

mary flow. Both efficiencies were calculated for a work output of 17.00 Btu per pound (39 572 J/kg), based on the primary mass flow and the overall total-pressure ratio. The efficiency of this turbine at zero coolant fraction was 0.900. Figure 13 shows that the thermodynamic efficiency of the turbine decreased significantly with coolant fraction. For example, at a coolant fraction of 0.07, the turbine efficiency was down about 0.09.

It has been shown that turbine overall performance was adversely affected by the transpiration-cooled stator. It was considered of interest, then, to determine whether rotor performance was also affected by the stator cooling air. Rotor efficiency values were calculated from experimental results using the following equation:

$$\eta_r = \frac{\Delta h_r}{\Delta h_{id,r}} = \frac{\dfrac{\tau N \pi}{(w_p + w_c)30J}}{\dfrac{V_1^2}{2gJ} + (h_1 - h'_{2,id})} \tag{5}$$

which expresses the specific-work output of the total flow through the rotor to the ideal rotor work. This ideal work, as used herein, is equal to the kinetic energy of the flow from the stator plus the ideal enthalpy drop across the rotor corresponding to the measured (and calculated) static-pressure to total-pressure drop across the rotor p_1/p'_2.

The kinetic energy from the stator $V_1^2/2gJ$ was determined from measured pressures and the experimental stator results given in reference 7. The calculated rotor efficiency at zero coolant flow was 0.930. Changes in the calculated rotor efficiency with coolant addition are shown in figure 14. The curve, normalized to the zero coolant flow condition, shows a drop of 1 percent with coolant fractions to 0.04; at a coolant fraction of 0.07, the drop in rotor efficiency was 2 points. This 2-point drop can be related to the previously mentioned 9-point drop in turbine thermodynamic efficiency at the 7-percent-cooling-air condition (see fig. 13) indicating the stator contributed to most of the turbine inefficiency.

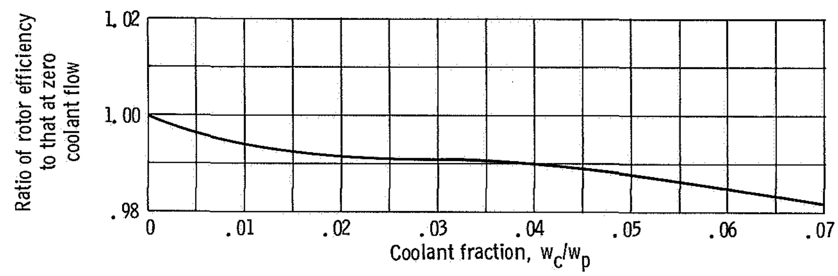

Figure 14. - Rotor efficiency calculated from turbine performance data and stator tests of reference 7. Equivalent design speed; equivalent work output, 17.00 Btu per pound (39 572 J/kg), based on primary flow.

The high stator loss resulted from a thickening of the boundary layer caused by the interaction of the coolant and primary air along the suction, or convex, surface of the blade, as reported in reference 7. These wakes at the stator exit resulted in a nonuniform, circumferential velocity profile entering the rotor, which further reduced turbine efficiency from that indicated from stator tests. From an aerodynamic standpoint, then, it can be concluded that the subject transpiration-cooled stator blades resulted in a severe penalty to overall turbine performance.

Comparison of performance between turbines with stator-trailing-edge coolant ejection and transpiration-cooled stator blades. - The turbine thermodynamic efficiencies obtained with trailing-edge-cooled and transpiration-cooled stator blades as functions of coolant fraction are compared in figure 15. Over the range of coolant fraction investigated, the efficiency of the transpiration-cooled turbine was less than that of the turbine with stator-trailing-edge coolant ejection.

As stated previously, the lower efficiency of the transpiration-cooled turbine was caused primarily by stator losses encountered with this cooling scheme. At zero coolant flow, the efficiency of the transpiration-cooled turbine was 0.900. This efficiency was about 2 points less than the efficiency for the slotted-stator-trailing-edge turbine. With no coolant flow, the increased stator loss was caused by primary air leakage

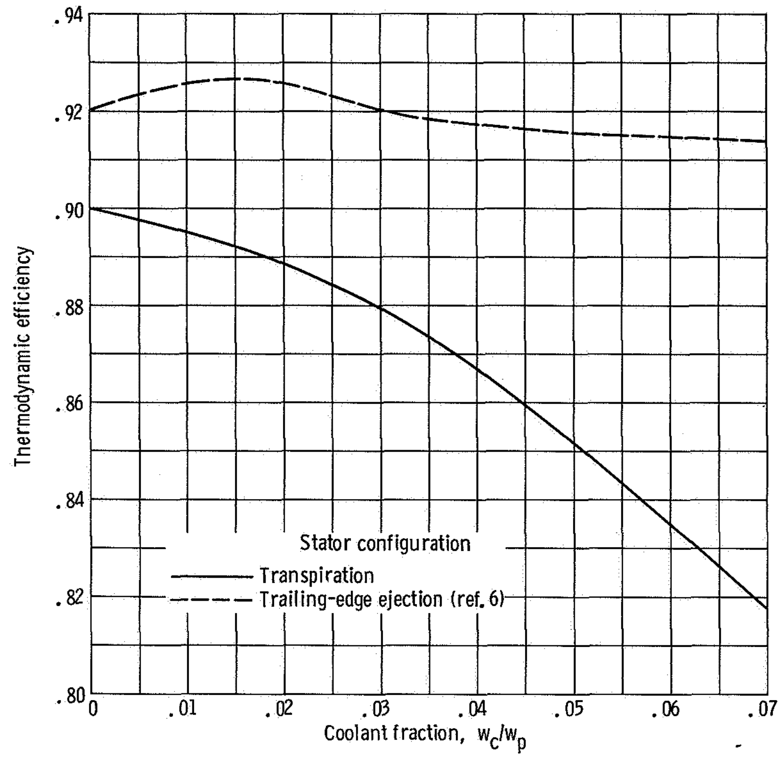

Figure 15. - Comparison of turbine thermodynamic efficiency for two different cooled-stator configurations. Equivalent design speed; equivalent work output, 17.00 Btu per pound (39 572 J/kg), based on primary flow.

through the cooling holes from the blade pressure side to the suction side. With coolant flow, the higher stator loss was caused by the direction and location of the coolant ejection relative to the primary flow. At a coolant fraction of 0.03, the efficiency of the transpiration-cooled turbine was 0.879 as compared to 0.920 for the slotted-trailing-edge turbine. At a coolant fraction of 0.07 (the maximum tested), the efficiency of the transpiration-cooled turbine was 0.818 as compared to 0.914 for the other. These results show that for the type of stator transpiration cooling investigated, the available energy of the coolant flow was not efficiently utilized to increase the overall turbine work output.

In marked contrast to the transpiration-cooled stator, the thermodynamic efficiency of the turbine with stator-trailing-edge coolant ejection was little affected by coolant flow (fig. 15). In fact, at low coolant fractions, the efficiency was improved by a slight amount. This small improvement results from a reduction in the stator-trailing-edge loss as the coolant filled the wakes behind the stator. As the coolant fraction was increased, the energy of the coolant flow (inlet pressure) became higher relative to the primary air. The leveling off of the thermodynamic efficiency at higher coolant fractions indicated the coolant flow was being utilized almost as efficiently as the primary flow. From these results it can be concluded that turbines using transpiration-cooled stator blades of the type reported herein are less efficient than turbines utilizing stator-trailing-edge-slot coolant ejection. It should be reiterated that all turbine tests were conducted using cold-air turbine inlet conditions. No heat transfer or coolant-passage loss effects were considered.

SUMMARY OF RESULTS

A 30-inch (0.762-m) single-stage turbine equipped with stator blades of a porous material with discrete holes was tested. The purpose of the test was to determine the effect of coolant flow ejection through this type of blade on turbine aerodynamic performance. The effect of coolant flow ejection was investigated by testing over a range of coolant fraction from zero to 0.07. The performance of this turbine was compared to a turbine utilizing stator-trailing-edge coolant flow ejection. Also, a comparison was made to a base reference turbine with no coolant flow provisions. Inlet total-state conditions for all three turbines were 30.0 inches of mercury absolute (10.159 N/cm^2) and approximately 545° R (303 K). The following results were obtained:

1. With a coolant supply pressure equal to 31 inches of mercury absolute (10.498 N/cm^2), the primary efficiency obtained at equivalent design speed and at a work output of 17.00 Btu per pound (39 572 J/kg) was 0.907 at a pressure ratio of 1.771. The corresponding coolant fraction was 0.0326.

2. The thermodynamic efficiency, which relates the torque output to the ideal energies of both the primary and coolant flows, decreased in almost direct proportion with coolant fraction. At a coolant fraction of 0.07, the thermodynamic efficiency was reduced about 9 percent. The primary air efficiency of the turbine, which relates the torque output to the ideal energy of only the primary flow, increased slightly with coolant fraction, reaching a maximum improvement of 1 percent at a coolant fraction of 0.07. The low efficiency encountered in the transpiration-cooled turbine was caused primarily by the high stator losses encountered with this type of cooling scheme. These high stator losses were caused by the location and direction of the coolant flow relative to the primary flow.

3. The high stator losses with the subject transpiration-cooled turbine were amplified in the rotor, causing a reduction of 2 points in rotor efficiency at a coolant fraction of 0.07. This additional loss was attributed to the nonuniform, circumferential velocity profile from the stator.

4. Both the primary and thermodynamic efficiencies of the turbine with transpiration-cooled stator blades were significantly less than the corresponding efficiencies obtained with stator-trailing-edge coolant ejection. For example, at a specific-work output of 17.00 Btu per pound (39 572 J/kg), design speed, and a coolant fraction of 0.07, the

thermodynamic efficiency of the transpiration-cooled turbine was 0.09 less than at a coolant fraction of zero. The thermodynamic efficiency of the turbine with trailing-edge coolant ejection remained essentially constant over the coolant range investigated.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, July 21, 1970,
720-03.

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